Prototype Vent Gas Heat Exchanger for Exploration EVA – Performance & Manufacturing Characteristics

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NASA is developing new portable life support system (PLSS) technologies, which it is demonstrating in an unmanned ground based prototype unit called PLSS 2.0. One set of technologies within the PLSS provides suitable ventilation to an astronaut while on an EVA. A new component within the ventilation gas loop is a liquid-to-gas heat exchanger to transfer excess heat from the gas to the thermal control system's liquid coolant loop. A unique bench top prototype heat exchanger was built and tested for use in PLSS 2.0. The heat exchanger was designed as a counter-flow, compact plate fin type using stainless steel. Its design was based on previous compact heat exchangers manufactured by United Technologies Aerospace Systems (UTAS), but was half the size of any previous heat exchanger model and one third the size of previous liquid-to-gas heat exchangers. The prototype heat exchanger was less than 40 cubic inches and weighed 2.57 lb. Performance of the heat exchanger met the requirements and the model predictions. The water side and gas side pressure drops were less 0.8 psid and 0.5 inches of water, respectively, and an effectiveness of 94% was measured at the nominal air side pressure of 4.1 psia.

I. Nomenclature

acfm	=	actual cubic feet per minute
lb	=	pounds
psia	=	absolute pressure in pounds per square inch
psid	=	differential pressure in pounds per square inch
Px	=	pressure transducer
Tx	=	thermocouple
Vx	=	valve

I. Introduction

As the United States looks to once again explore beyond low Earth orbit, the technologies necessary for the effort must be available to support the endeavor. One of these technologies is the Portable Life Support System (PLSS) for a future exploration space suit. While the PLSS on NASA's Extravehicular Mobility Unit (EMU) has successfully served our astronauts for over thirty years, the requirements necessary for missions to the moon, Mars or an asteroid preclude its use. In order to meet the need for a new PLSS, NASA Johnson Space Center (JSC) began incremental development of PLSS technologies in 2011 with their PLSS 1.0 breadboard test bed[(1)]. Results and lessons learned from PLSS 1.0 were used to design PLSS 2.0, which incorporates higher fidelity components in a packaged volume similar to what is anticipated for a new flight PLSS. Differences in the ventilation loop schematic between PLSS 2.0 and the EMU PLSS have resulted in gas flow that no longer gets chilled in a condensing heat

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exchanger. To compensate for the change, NASA added a small gas to water heat exchanger into their schematics to cool the ventilation flow[2].

United Technologies Aerospace Systems (UTAS) was contracted to design and build the custom gas to liquid heat exchanger for NASA's PLSS 2.0. The challenge centered on achieving 85% effectiveness in a compact, lightweight design that had minimal pressure drop on the gas side and was corrosion resistant on the water side. The original design requirements are shown in Table 1. During the program, the mass requirement was loosened to 2.0 lb, not including the compression fittings. The resulting design was 50% smaller than any heat exchanger UTAS has designed or made, which added uncertainty to our sizing analysis and risk to the manufacturing process. Special considerations were successfully made to accommodate these unique aspects of the heat exchanger.

Table 1: Heat Exchanger Design Requirements

Heat Exchanger Requirements	
Mass:	≤ 1.5 lbm
Volume:	≤ 40 in^3
Effectiveness:	≥ 85%
Collapse Pressure:	Unit must withstand up to 13 psid; between both loops and both the environment and internal volume and any internal differential
Structural Margin:	at least 2.0 times burst/collapse pressure
Liquid Side (Water) Specifications	
Flow Rate:	200 lbm/hr (91 kg/hr)
Pressure Drop:	< 0.8 psi @ 200 lbm/hr
Inlet Temp:	55°F +/- 5°F
Inlet Pressure:	3.3 - 55 psia
Max-Pressure:	Unit must withstand up to 40 psid; between both the environment and internal volume and any internal differential
Leakage:	**Must be bubble tight at MDP**
Inlet and Outlet Ports:	3/8" male AN
Gas Side (Nitrogen) Specifications	
Flow Rate:	6 +/-.5 acfm
Pressure Drop:	< 0.5 inches of water @ 6.0 acfm and 4.1 psia
Inlet Temp:	90°F ± 15°F
Inlet Pressure:	3.3 psia-25 psia
Max-Pressure:	>9 psid; between both the environment and internal volume and any internal differential
Leakage:	Pressurize to 25 psia, leakage shall not exceed 10 scc/hr
Inlet and Outlet Ports:	3/4" male AN

II. Analysis and Design Description

A. Requirements

The PLSS heat exchanger sizing process utilized a Hamilton Sundstrand custom plate-fin analytical model to define the optimum (minimum weight) design. The model utilized a multivariable solver routine that varied overall heat exchanger envelop, fin layer count, and fin dimensions to reach a solution. The solver was configured to meet all design constraints and performance requirements at the specified design point conditions. Table 2 shows the relevant design point conditions and target requirements.

Table 2 – Design Point Requirements

Parameter	Low Pressure Design Point	High Pressure Design Point
(Interface) Hot Fluid Type	Nitrogen	Nitrogen
(Interface) Hot Flow	6.5 acfm (7.4 lbm/hr)	6.5 acfm (45.1 lbm/hr)
(Interface) Hot Inlet Pressure	4.1 psia	25.0 psia
(Interface) Hot Inlet Temperature	105°F	105°F
(Interface) Cold Fluid Type	Water	Water
(Interface) Cold Flow	200 lbm/hr	200 lbm/hr
(Interface) Cold Inlet Temperature	60°F	60°F
(Constraint) HX Effectiveness	**≥ 85%**	**≥ 85%**
(Constraint) Hot Pressure Drop	**≤ 0.5 inH_2O**	**-**
(Constraint) Cold Pressure Drop	**≤ 0.8 psid**	**-**
(Constraint) Hot Duct I.D.	**0.63 in**	**0.63 in**
(Constraint) Cold Duct I.D.	**0.31 in**	**0.31 in**

For these conditions, the heat exchanger design was limited thermally by the high pressure case due to its much higher nitrogen mass flow rate. Conversely, the limiting pressure drop case was at the low pressure due to the higher velocities. Both conditions were accounted for during the design process.

B. Design Selection

The selection of heat exchanger type (plate-fin, shell-tube, etc.) is typically driven by some combination of performance, cost, weight/volume, and structural limitations. For this application, the plate-fin heat exchanger type was selected due to (1) high performance per unit volume and (2) company experience manufacturing these devices. Typically, liquid-to-gas heat exchangers tend towards a cross-flow configuration due to large disparities in velocity and allowable pressure drop between fluids. However, the high water-to-nitrogen mass flow ratio in this unit allowed for the use of a high effectiveness counterflow configuration. This resulted in a smaller, more compact core design.

C. Header Design

The nitrogen side header configuration provided a unique performance challenge for this heat exchanger. Typically, the interface duct diameters are sized such that the duct velocity head is relatively low compared to core (fin) pressure drop. This design guideline ensures uniform flow distribution through the core fin layers. However, due to interface constraints the header velocity head was more than two times the core pressure drop. In order to address this issue, the heat transfer area was sized with extra margin and worst case fin tolerances were applied to the model.

III. Hardware

A brazed, compact heat exchanger approach was used for the unit, with 347 stainless steel and nickel-based braze alloy as the primary materials. The headers were machined parts, rather than welded assemblies because of the small size of unit. Flared AN fittings were specified by NASA, and were machined from 347 stainless steel for welding and for corrosion resistance. The headers were passivated and then welded onto the brazed core. Figure 1 is a photograph of the final item.

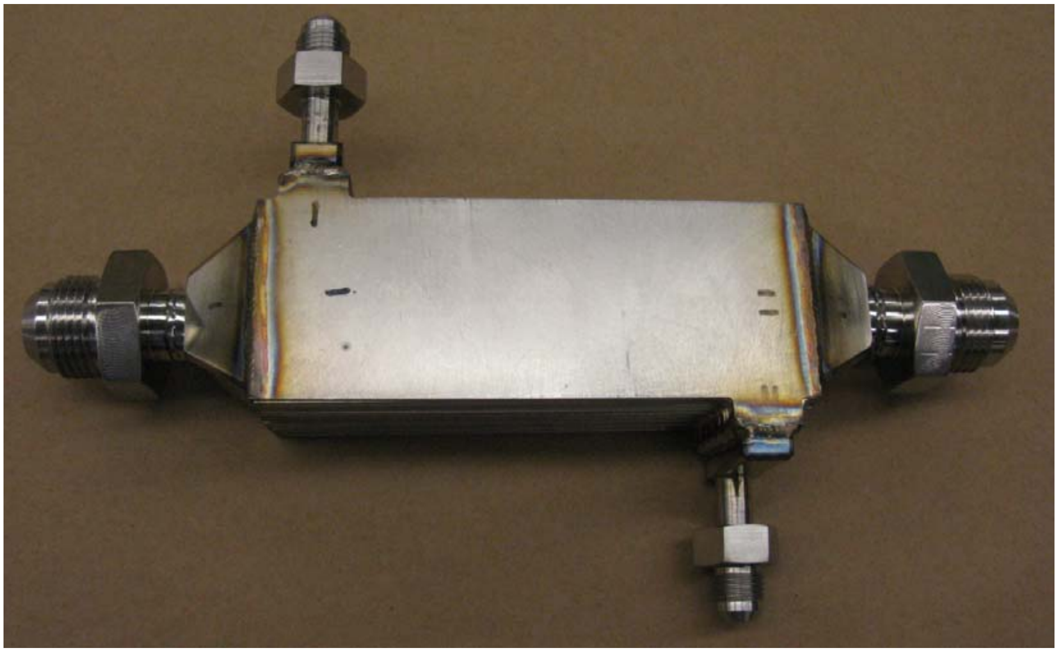

Figure 1: Brazed and welded PLSS heat exchanger

The nitrogen side of the heat exchanger has a maximum design pressure (MDP) of 25 psig and was proofed and leak checked at 37.5 psig, while the water side of the heat exchanger has a maximum design pressure of 55 psig and was proofed and leak checked at 82.5 psig. Leak checks were conducted by pressurizing one set of passages to 1.5 x MDP with nitrogen, submersing the item under water and checking for visible bubbles on the outside of the item. Inter-path leakage was checked at the same time by looking for bubbles at the exit of the unpressurized headers. No leaks were found.

Among the requirements for mass, volume and effectiveness, the effectiveness and the volume were considered the most important for PLSS 2.0. Early iterations of the heat exchanger analysis indicated that the performance would be best met with a heat exchanger that weighed slightly more than 1.5 lb (exclusive of fittings). As a result, the as-built heat exchanger with headers and fittings is 2.57 lb. Estimates for the mass of the four AN fittings was 0.57 lb, resulting in a heat exchanger mass and header mass of 2.0 lb.

The heat exchanger volume without fittings met the requirements of 40 cubic inches. It fits in a prismatic envelope measuring 6.87 inches x 1.78 inches x 3.2 inches, for an overall volume of 39.1 cubic inches. A solid model of the unit was provided to NASA for PLSS 2.0 packaging design work, which allowed the actual heat exchanger to easily integrate into the PLSS after it was delivered.

IV. Test Setup & Instrumentation

Performance testing was conducted on the PLSS heat exchanger to determine its pressure drop and effectiveness at the nominal and worst case design conditions. Figure 2 shows the schematic of the test setup for sub-ambient gas pressure and Figure 3 shows the schematic for above-ambient gas pressure. The heat exchanger and headers were insulated in all tests. However the first set of tests indicated that heat was leaking into the lines from the room, particularly around thermocouple T2, which provided a key measurement for calculating the heat exchanger's effectiveness. As a result, the measured outlet gas temperature was high in tests 1-7, and resulted in a slight under-reporting of the effectiveness for that set of tests. These tests were not all repeated, though, because all of them met the performance requirements. Additional insulation from the thermocouples up to the heat exchanger was added for tests 8-11.

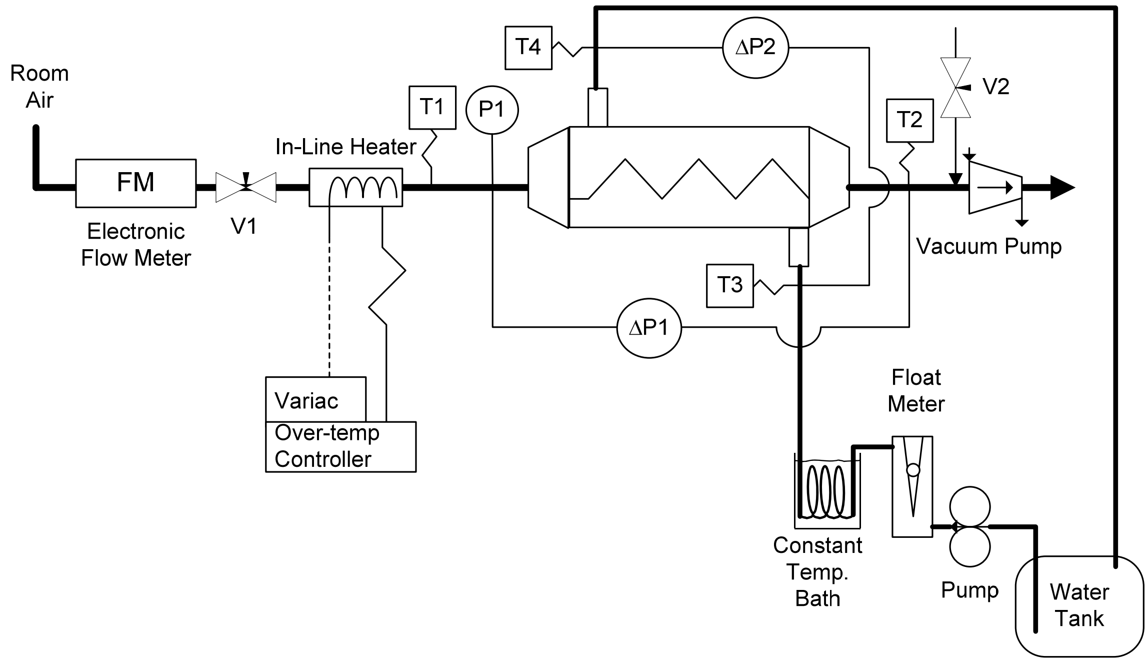

Figure 2: Heat exchanger test schematic for sub-ambient conditions

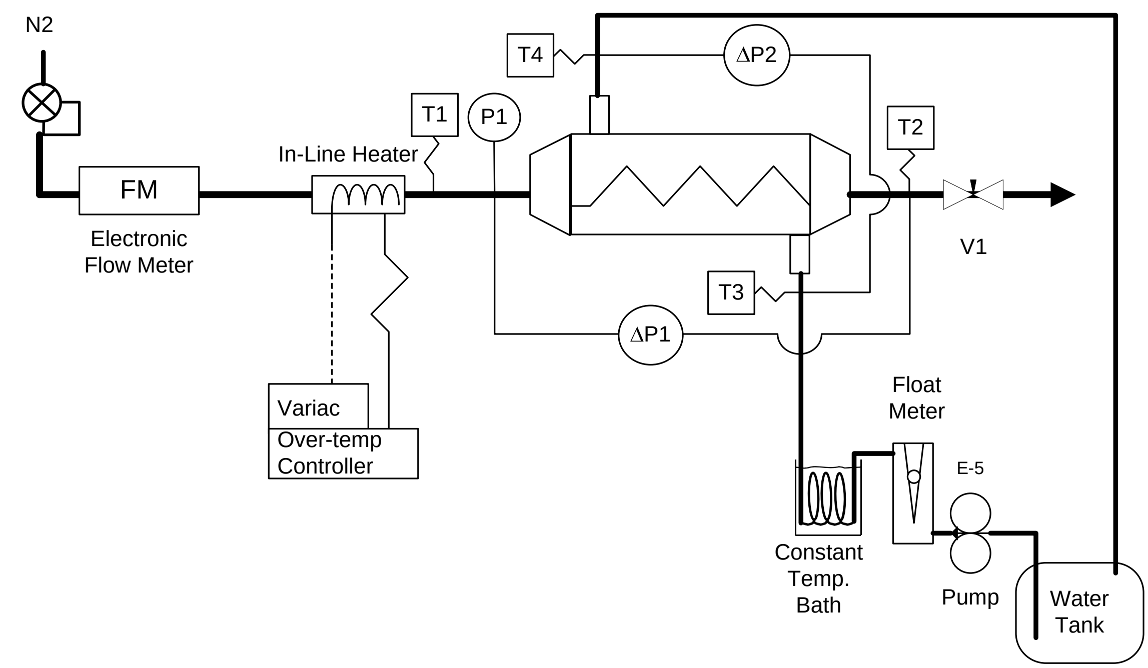

Figure 3: Heat exchanger test schematic for above-ambient conditions

V. Procedure

Table 3 contains the test matrix and results. Tests were conducted in order to verify the effectiveness and pressure drop of the heat exchanger. For tests with sub-ambient gas pressure, the vacuum pump pulled ambient air through the setup and heat exchanger. Pressure was set by adjusting valve V2 and air flow was set by adjusting valve V1. For tests with above-ambient gas pressures, the nitrogen flow rate and pressure in the heat exchanger was controlled by adjusting the nitrogen supply regulator and valve V2, respectively. Gas inlet temperature was set by adjusting the heater power via the variac. The water inlet temperature was set by

adjusting the bath temperature. Water flow rate was controlled using a dial potentiometer on the positive displacement pump.

Test points were held for a minimum of five minutes to ensure that steady state was achieved in the heat exchanger. Three test points, 8, 10 and 11, were run for 10 to 20 minutes, which verified that steady state had actually been achieved in the remaining tests.

The gas-side heat exchanger pressure drop measurements included the pressure drop through several inches of small fittings. A "Tare" was done on the fittings for the sub-ambient test conditions by repeating the measurements with a short tube in place of the heat exchanger. Measurements were done with the same pressure, flow rate and inlet temperature as the test points shown in Table 3. The reported pressure drop is adjusted by subtracting out the fitting losses on both the gas and the water side. Measurements of the gas side fitting losses at 14.9 psia and 25 psia were not made, and the raw pressure drop values are reported as an upper bound for tests 5-9.

Table 3: Test Matrix and Results

Test Point	Gas side					Water side				Effectiveness	Comments
	Inlet Temp	Outlet Temp	Pressure	ΔP	Flow Rate	Inlet Temp	Outlet Temp	Flow Rate	ΔP	Measured	
	°F (±.9)	°F (±.9)	Psia (±0.3)	inches H2O (±0.05)	ACFM (±0.3)	°F (±.9)	°F (±.9)	lb/hr (±1)	Psid (±.04)	% (±2)	
1	105	58	4.3	**0.11**	5.7	55	55	197	**0.36**	**94%**	Nominal pressure, high air temp, nominal flow
2	90	57	4.3	**0.11**	5.7	55	55	198	**0.35**	**93%**	Nominal pressure, nominal air temp, nominal flow
3	90	57	4.2	**0.11**	6.3	55	55	198	**0.41**	**94%**	Nominal pressure, nominal air temp, high flow
4	90	57	4.2	**0.15**	5.4	55	55	196	**0.42**	**94%**	Nominal pressure, nominal air temp, low flow
5	75	58	14.9	**< 6.68**	5.9	55	56	197	**0.44**	**85%**	Ambient pressure, low air temp, nominal flow
6	90	59	14.9	**<6.68**	5.9	55	56	197	**0.44**	**90%**	Ambient pressure, nominal air temp, nominal flow
7	105	60	14.9	**<6.74**	5.9	55	57	196	**0.44**	**91%**	Ambient pressure, high air temp, nominal flow
8	76	58	25.0	**<13.0**	6.2	54	55	196	**0.34**	**84%**	High pressure, low air temp, nominal flow
9	90	59	25.0	**<13.2**	6.2	54	56	196	**0.34**	**85%**	High pressure, nominal air temp, nominal flow
10	91	56	4.1	**0.15**	6.0	54	55	196	**0.43**	**95%**	Nominal pressure, nominal air temp, nominal flow
11	75	55	4.1	**0.19**	6.0	54	54	197	**0.43**	**95%**	Nominal pressure, low air temp, nominal flow

VI. Results & Discussion

Results are shown in Table 3 and the uncertainties shown were calculated using the Kline-Mclintock method. All but one of the tested conditions yielded calculated effectiveness values that meet the requirement of 85%. Nominal conditions were represented by test points 2 and 10 and yielded heat exchanger effectiveness values of 93% and 95%, respectively. Pressure drop on the air side for these two tests was 0.11 inches of water for test 2 and 0.15 inches of water for test 10. The difference is likely due to the 0.3 acfm increase in flow from test point 2 to 10. All of the tests conducted at 4.1 psia yielded gas side pressure drops of less than 0.20 inches of water, which meets the requirement 0.5 inches of water maximum pressure drop. The “tare” pressure drop on the air side at 4.1 psia ranged from 1.4 to 2.0 inches of water. The water side pressure drop ranged from 0.35 to 0.44 psid, and met the requirement of 0.8 psid maximum pressure drop. The “tare” pressure drop on the water side ranged from 0.29 to 0.44 psid.

The one condition that yielded an effectiveness of less that 85% was the high pressure, low inlet temperature condition shown in test point 8, with a measured effectiveness of 84%. This was not surprising, because the high gas pressure conditions were predicted to be the most challenging for achieving the required 85% effectiveness. While the 75 °F inlet test at 25 psia just missed the mark, the 90 °F inlet test (test 9) resulted in an effectiveness of 85 %.

Figure 4 shows the trend of increasing effectiveness with decreasing gas pressure. This trend was predicted by our analytical model and is the reason why we sized the heat exchanger to meet the 85% effectiveness requirement at 25 psia gas pressure. Figures 5 and 6 show that the gas inlet temperature and the gas flow rate have no significant effect on the heat exchanger performance. This is because the properties of nitrogen and air do not change significantly with temperature within the specified range. The gas flow regime is also constant from 5.5 to 6.5 acfm so that the film coefficient and Nusselt Number of the heat exchanger do not vary significantly within that range. While the higher gas flow rates affected the NTU’s and the capacitance ratio, the effect is small with the NTU’s being over 5 and the capacitance ratio near zero in all cases. These small effects are somewhat counter-balanced by the reduced effect that lateral conduction has with the higher flow rate.

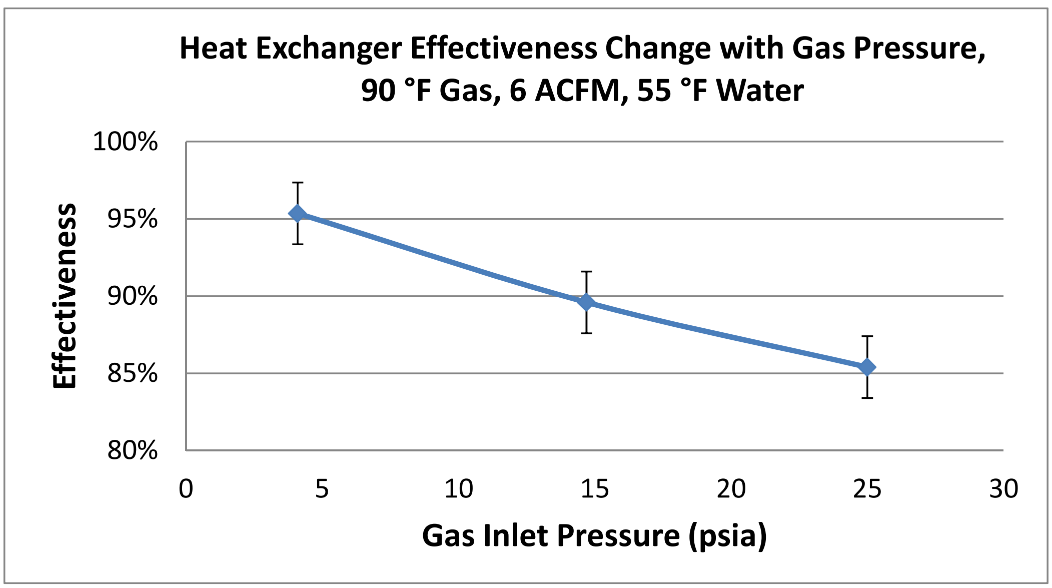

Figure 4: Higher inlet gas pressure reduces the effectiveness of the heat exchanger

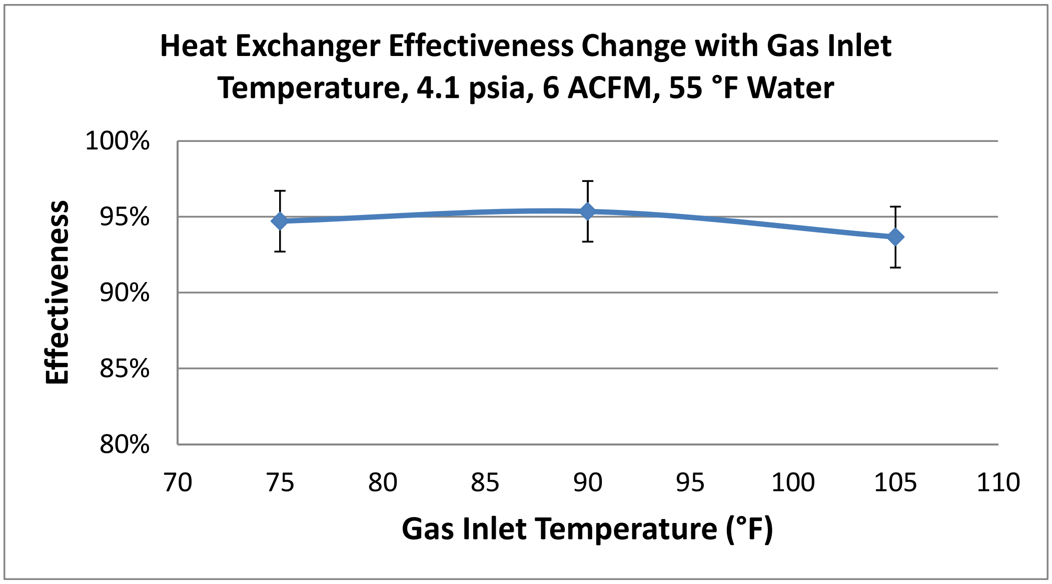

Figure 5: Gas inlet temperature does not affect heat exchanger performance

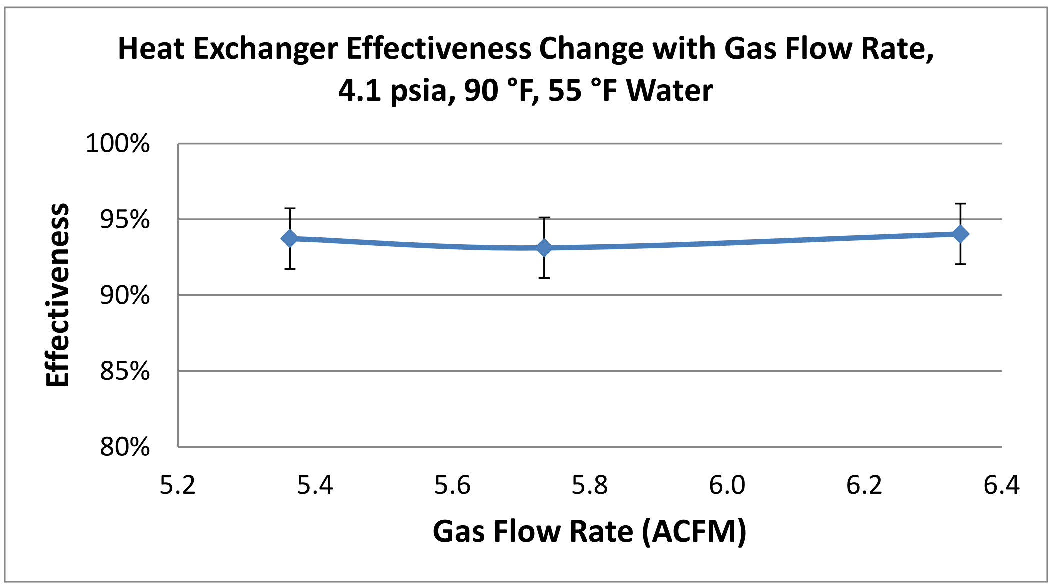

Figure 6: Gas flow rate does not affect heat exchanger performance within the tested range

VII. Conclusion

The PLSS gas to liquid heat exchanger met the leakage, volume and nominal performance requirements of the program. At 2.0 lb, the heat exchanger mass was 0.5 lb above the desired value, but acceptable for the current technology readiness of the overall PLSS. Performance of 85% effectiveness was met in all but one of the tests, which resulted in a calculated effectiveness of 84% ±2%. Pressure drop on the gas side at 4.3 psia was below 0.20 inches of water and well within the requirement of 0.5 inches of water. Pressure drop on the water was also below the limit of 0.8 psid, with a maximum pressure drop of 0.44 psid. These results point to some opportunities for reducing the heat exchanger size on future iterations of NASA's PLSS.

Acknowledgements

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References

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[2]Campbell, C., *Advanced EMU Portable Life Support System (PLSS) and Shuttle/ISS EMU Schematics, a Comparison*, AIAA-2012-3411, International Conference on Environmental Systems (ICES), San Diego, CA, July-2011.